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# An Assessment of Experimental Techniques for Measuring Fast Temperature Transients in Compressors

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## ABSTRACT

An experimental investigation has been carried out to measure gas temperature transients in the suction chamber of a small reciprocating compressor operating at 3600 rpm. To this extent, two different measurement techniques, represented by a hot-wire system and a micro-thermocouple, were applied. Results for valve displacement and pressure pulsation in the suction chamber were also obtained to complement the analysis. Measurements indicated abrupt variations in all flow properties, strongly linked to the valve motion. It has also been observed a considerable increase in the gas temperature during the period in which the suction valve is closed, associated with high heat transfer rates between the gas and the suction chamber walls.

## 1. INTRODUCTION

In household refrigeration compressors, a significant portion of the energy losses is associated with refrigerant superheating along the suction path. This useless superheating provokes a reduction in the volumetric efficiency and an increase in the compression work per unit mass. A detailed thermal analysis of the suction system can lead to a considerable improvement in the compressor performance, mainly through the reduction of gas superheating. However, a complete analysis of the involved phenomena requires also data for gas temperature transient, which is very difficult to obtain because the time scale of such variations is similar to that found in compressor valves.

The micro-thermocouple is a small size sensor capable of measuring fast transients without causing major interferences to the flow. Because micro-thermocouples only require simple auxiliary equipments and their calibration curves are available in the literature, they are very attractive when compared with other measuring techniques. For instance, the cold-wire anemometry is normally employed to measure temperature fluctuations in turbulent flow and, although its response time is much faster than that of micro-thermocouples, it requires costly equipments and may introduce some perturbations into the flow.

Lee and Smith (1980) measured inlet and outlet temperatures both inside the cylinder and outside the cylinder near the valves, with thermocouple probes fabricated with chromel-alumel wires of 25.4 $\mu$ m diameter. The probe assembly was designed to have a fast response (less than 1ms time constant). Prasad (1992) built a 12.7 $\mu$ m diameter chromel-constantan wire micro-thermocouple especially designed for temperature measurements in a two stage, double acting, compressor running at approximately 900rpm. He successfully measured the instantaneous temperature inside the cylinder and found a considerable gas superheating.

Experiments of flow in inlet manifolds of IC engine were conducted by Bauer *et al.* (1998) with the purpose of analyzing heat exchange for different flow parameters. Thermocouples were positioned in 5 positions along the manifold and four heat flux sensors in each intermediate section. Next to the exit section, the port was arranged to

fit, alternatively, a hot-wire, a cold-wire and a heat flux sensor. The cold-wire probe calibration was made to allow temperature measurements up to 100°C, with a correction for thermal inertia using a fixed time constant of 2ms, derived from energy balance and heat transfer correlations for the wire sensor. The authors concluded that the fluid in the manifold is strongly heated even when the valves are closed.

Zend and Assanis (2003) analyzed heat transfer in inlet manifolds of IC engines, pointing out that during the operation cycle the flow is composed of two distinct phases, one associated with the period in which the admission valve is open and the other in which the valve is closed. The authors developed a new heat transfer correlation to characterize the transient nature of the flow.

In an experimental investigation of flow through inlet and discharge manifolds of diesel engines, Olczyk (2008) identified that the main difficulties in measurements of temperature transient are related to both the sensor response time and the signal interpretation. The authors adopted thermocouples, hot-wires and cold-wires in an apparatus capable of generating pulsating flows with frequencies up to 180 Hz.

Seeking to characterize superheating in compressors, Morriesen and Deschamps (2009) measured the instantaneous temperature inside the suction chamber of a small reciprocating compressor operating at 3600 rpm. The authors chose an anemometry system with a single 5µm diameter tungsten wire sensor to measure velocity and temperature. A sudden temperature decrease was found during the opening of the suction valve, followed by a significant increase in the gas temperature for the period in which the valve is closed.

From the literature review, it is clear that few studies have applied single point measurement techniques, such as cold-wires and micro-thermocouples, to measure thermal transients in compressors. The main objective of this paper is to compare the capabilities of micro-thermocouples and cold-wire anemometers for measurements of fast temperature transients in the suction chamber of a small reciprocating compressor. Such an assessment is complemented with experimental data for valve displacement and pressure inside the suction chamber.

## 2. EXPERIMENTAL SETUP AND PROCEDURE

A reciprocating compressor operating with R134a was selected for the analysis, being submitted to two operating conditions, represented by evaporation and condensation temperatures: -23.3°C/54.4°C and -35.0°C/54.4°C. The first operating condition defines the pressure in the suction and discharge lines as 1.149bar and 14.71bar, respectively. When the evaporation temperature is lowered to -35.0°C, the suction pressure becomes equal to 0.662bar.

Piezoelectric transducers were selected for the pressure measurements in the suction chamber, due to their high response time, small size and reliability regarding the hostile conditions inside the compressor. To correlate the pressure measurement with the crank angle, a sensing winding was assembled to the crankcase to collect the signal emitted by a magnet fixed to the crankshaft. The instantaneous crankshaft position is calculated taking into account the compressor mechanism characteristics. Small sensing windings were also assembled into the valve plate seat to give the valve lift according to the crankshaft position.

Measurements of temperature were conducted by using a cold-wire probe and a micro-thermocouple probe. The cold-wire probe was composed by a DANTEC anemometer system with a constant current module and a single 5µm diameter tungsten wire sensor, which allowed measurements of velocity and temperature. The sensor was calibrated for velocity measurements by using a DANTEC calibration unit and a thermistor to register any significant variation of temperature during the process.

It should be mentioned that a number of aspects may affect measurements of velocity and temperature with wire sensors. For instance, Weiss *et al.* (2005) have shown that the presence of oil droplets in the flow can increase the sensor diameter with running time, leading to a reduction of its response time. Such a contamination process requires testing and cleaning the wire steadily in order to keep its response time. This is particularly relevant in compressors because oil is always present in the suction system.

The micro-thermocouple was an OMEGA model CHAL-0005, chromel-alumel with a wire diameter of 12.5µm. Following Lee and Smith (1980) and Prasad (1992), a special probe was conceived for the micro-thermocouple

junction, allowing its assembly into the suction chamber. Essentially, the probe has an exposed loop of the bare wire thermocouple and each one of lead wires passed through the two holes of a ceramic tube (Fig. 1). The ceramic tube was internally glued to a machined M6 screw, allowing the probe to be fastened to the suction chamber wall.

As illustrated in Fig. 2a, the anemometry sensor was assembled at the entrance of the suction chamber, allowing measurements of velocity and temperature. A small tap hole (1mm diameter) was used to connect the suction chamber to a larger orifice (7mm diameter), in which the piezoelectric pressure transducer was mounted. An additional muffler was prepared for measurements with the micro-thermocouple probe, which was also positioned at the entrance of the suction chamber (Fig. 2b). However, due to its size, the micro-thermocouple probe had to be fixed completely level with the surface and, hence, the junction was very close to the wall.

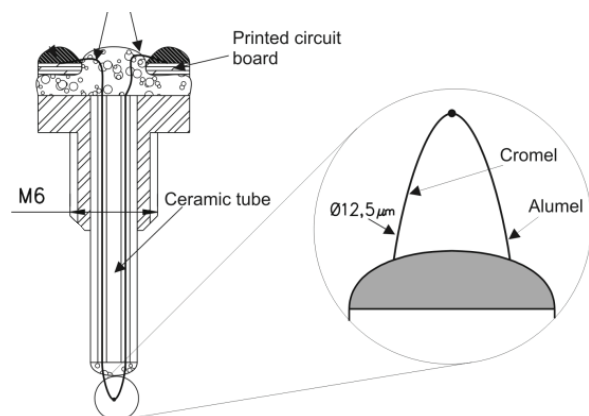


Figure 1: Micro-thermocouple probe.

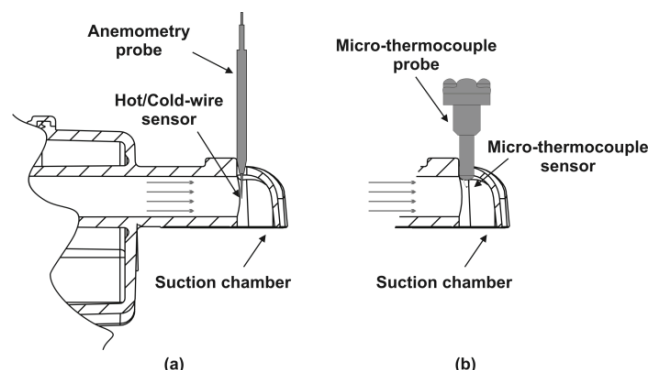


Figure 2: Assemblies of the anemometry and micro-thermocouple probes into the suction chamber.

A calorimeter was employed to investigate the compressor under the specified operating conditions. The uncertainties associated with measurements taken with the calorimeter are  $\pm 3\%$  for mass flow rate and power consumption. Further details of the experimental facility can be found in Pereira *et al.* (2008). The acquisition system was composed by a computer, a National Instruments converter analogue/digital (A/D), and a program for data acquisition and reduction, developed with LabVIEW 8.6. The compressor was tested three times for each operating condition.

The first step in the experimental procedure is to submit the compressor to an adequate vacuum condition, in order to remove air, humidity and any other contaminant inside the system. Then, the system receives a charge of refrigerant and the flowmeter reading is set to zero. After the compressor is switched on, a period of approximately 2 hours is needed to establish a fully periodic operating condition because of compressor thermal inertia. Finally, data for temperature, velocity and pressure in the suction chamber and displacement of suction valve are collected for 240 operation cycles of the compressor.

A total of 1000 measurements points were acquired per cycle to properly characterize the compressor. Since the compressor operates at a velocity of 3600 rpm, an acquisition rate of 60kHz was adopted. The 240,000 experimental data collected along 240 cycles were statistically analyzed, with ensemble average results being expressed as a function of the crankshaft angle. Average data for energy consumption and mass flow rate were acquired during the compressor operation.

### 3. RESULTS AND DISCUSSION

Figure 3 shows velocity in the suction chamber as a function of the crank angle measured by the anemometry sensor for the operating condition  $-23.3^{\circ}\text{C}/54.4^{\circ}\text{C}$ . Data for pressure and valve displacement are also shown to assist the analysis. The valve opens at the crank angle  $230^{\circ}$ , followed by a pressure drop and a velocity increase that reaches approximately 30m/s at  $264^{\circ}$ . It is interesting to note that points of maximum for velocity and valve displacement do not occur at the

same crankshaft angle. Firstly, the hot-wire sensor was installed at the entrance of the suction chamber and the fluid has to flow along a distance before reaching the suction port, which contributes for a time delay between data of velocity and valve displacement. Naturally, the other aspect taking part in this delay is flow inertial effects. Despite the difficulty originated by the two aforementioned aspects, it is clear that the velocity reaches its maximum value when the valve is fully open.

When the valve is closed, the anemometry sensor indicates velocity oscillations with amplitudes approximately equal to 3m/s. This phenomenon is associated with expansion and compression pressure waves traveling forward and backward in the duct that feeds the suction chamber. Naturally, the hot-wire sensor is insensitive to the velocity direction and, therefore, cannot detect between outflow and inflow conditions in the suction chamber. For this reason, velocity magnitudes are always shown as positive in Fig. 3.

Figure 4 was prepared to closely analyze pressure and velocity in the suction chamber during the valve opening period. Subtle changes in the velocity time variation can be noticed around the crankshaft angles 250° and 276°, also observed in the raw experimental data, which are probably linked to flow perturbations. A phase delay is seen between the opening of the valve and the beginning of the flow in the duct, which is also probably due to the fluid inertia and the positioning of both sensors.

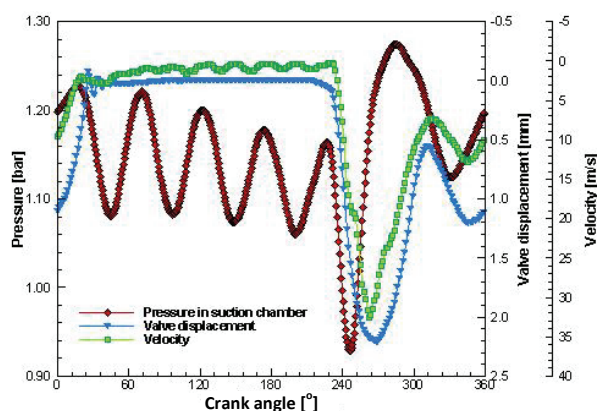


Figure 3: Measurements of velocity, pressure and valve displacement (-23.3°C/54.4°C).

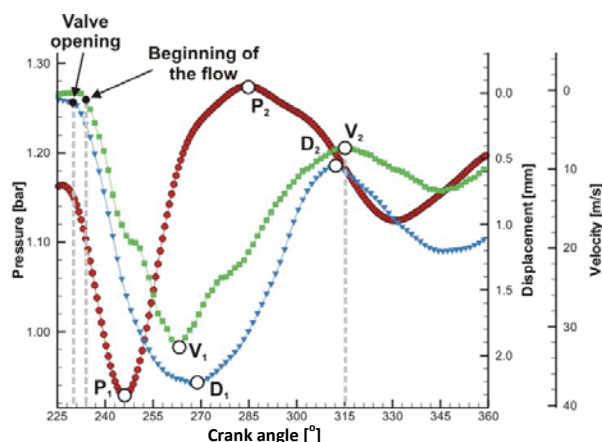


Figure 4: Results for velocity and pressure during the opening of the valve (-23.3°C/54.4°C).

Initially, there is a pressure decrease in the suction chamber because gas is supplied to the suction port. After reaching point  $P_1$ , pressure rises and the flow starts to decelerate, as indicated by the velocity reduction after the maximum  $V_1$  at the crank angle 262°. Finally, the pressure is observed to decrease again at the crank angle 285°, probably due to a greater mass flow rate exiting the suction chamber through the valve port.

After point  $D_2$ , the pressure field causes the valve to reverse its motion and this is followed by an increase of velocity in the suction chamber after point  $V_2$ . At the crank angle 330°, the suction chamber pressure starts to increase again and the velocity is then reduced as the valve approaches its closing point. As can be seen from Fig. 3, the valve closes at the angle 24°. Figure 5 depicts measurements of pressure, velocity and valve displacement for condition -35°C/54.4°C, showing similar phenomena observed for -23.3°C/54.4°C (Fig. 3).

Results for valve displacement are shown in Fig. 6 for the operating conditions -23.3°C/54.4°C and -35°C/54.4°C. It is observed a difference between phase and amplitude in the curves for both conditions. The phase shift of 19° can be explained by a delay in the valve opening for the condition -35°C/54.4°C. Table 1 shows the corresponding pressures in the suction line for both operating conditions. It is well known that before the suction valve can open and allow vapor to enter the cylinder, the high-pressure vapor left in the clearance volume at the end of the discharge process must be expanded to a pressure below that in the suction line. Since the discharge pressure is the same in both operating conditions, the clearance vapor must expand to a lower pressure in the case of a lower evaporation temperature. As a consequence, there is a delay in the opening of the suction valve for the condition -35°C/54.4°C, which occurs at the crank angle 245°. For this operating condition, the valve rises to a maximum displacement of

1.45mm at 286° and almost reaches the closing position at 340°. Connected to a late opening and small amplitudes of valve displacement, the compressor returns a lower mass flow rate (2.20kg/h), as indicated in Table 1.

Table 1. Compressor operating conditions adopted in the experiments.

Operating condition (°C/°C) <sup>(1)</sup>	-23.3/54.4	-35/54.4
Suction pressure (bar)	1.150	0.656
Discharge pressure (bar)	14.71	14.74
Mass flow rate (kg/h)	5.07	2.20
Average suction temperature (°C)	51.3	57.0

<sup>(1)</sup>: Evaporation/Condensation temperatures

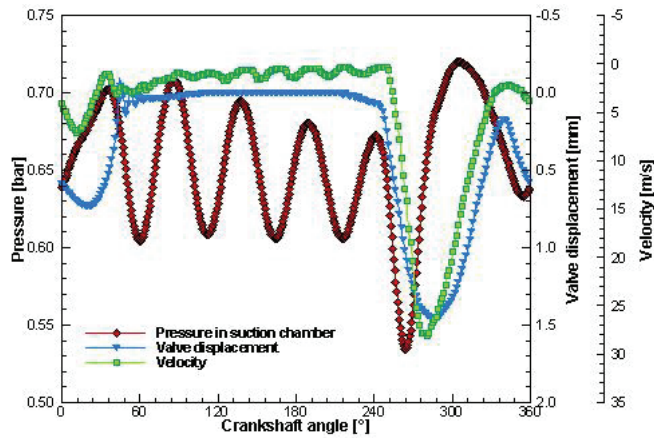


Figure 5: Measurements of velocity, pressure and valve displacement (-35°C/54.4°C).

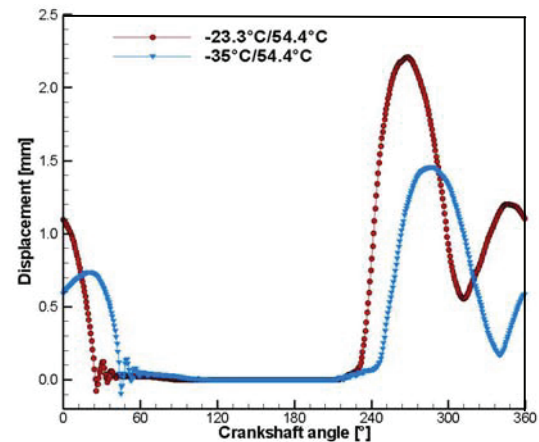


Figure 6: Valve displacement as a function of the operating condition.

Regarding pressure pulsation in the suction chamber, Fig. 7 shows a phase shift between the curves for both operating conditions, as should be expected. The maximum pressure in the second operating condition is 0.72bar at 304°, and the minimum level is 0.53bar at 264°. Measurements of velocity during the valve opening period (Fig. 8) indicate that values of maximum and minimum are lower for the condition -35°C/54.4°C, remaining with velocity levels between 2 and 7m/s during the second oscillation of the valve motion.

Figures 9 and 10 show temperature measurements in the suction chamber along the compressor cycle, obtained with the cold-wire anemometer and the micro-thermocouple for the conditions of -23.3°C/54.4°C and -35°C/54.4°C, respectively. Because measurements were carried out three times for each operating condition, data at each crank angle represents the arithmetic mean of three values.

As can be seen from Figs. 9 and 10, the cold-wire sensor shows a much faster response when compared with the micro-thermocouple, being capable of following temperature variations of 4°C/ms whereas the micro-thermocouple was seen to reach only 1°C/ms. Naturally, this is a consequence of the greater thermal inertia of the micro-thermocouple and, therefore, the cold-wire sensor seems to be more appropriate to characterize the fast temperature transients verified in the suction chamber of this compressor.

Mean temperature values can be evaluated in the suction chamber along one cycle, providing further information concerning measurement accuracy. Considering the condition -23.3°C/54.4°C, the mean temperature is found to be 51.2°C for the cold-wire and 51.4°C for the thermocouple, whereas for the other condition (-35.3°C/54.4°C) the values returned by each sensor are 57.9°C and 56.8°C, respectively. Hence, despite different capabilities to measure temperature transients, both sensors return similar results for mean temperature, as one would expect.



Figure 9 shows that the maximum and minimum temperature registered by the cold-wire sensor in the suction chamber are equal to 54.2°C and 47.8°C at the crank angles 237° and 339°, respectively. The temperature decrease observed when the valve opens can be attributed to the pressure drop in the suction chamber. For the short period of time (9ms) in which the valve is closed, there is a temperature increase of approximately 4°C in the suction chamber, suggesting high heat transfer rates between the gas and the chamber walls. Similar conclusions can be drawn from the results for the other operating condition (-23.3°C/54.4°C), as illustrated in Fig. 10.

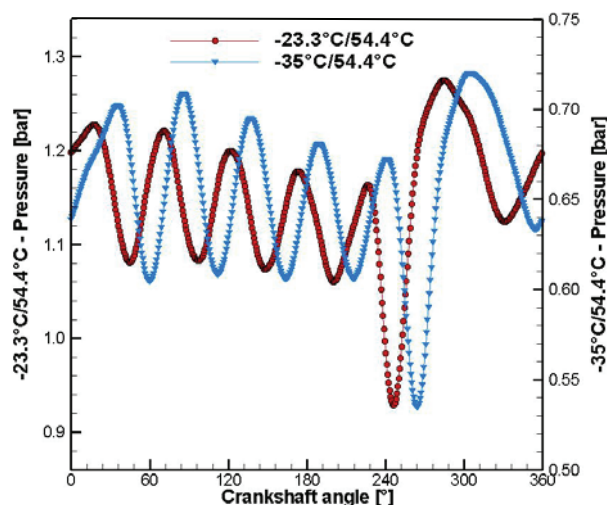


Figure 7: Pressure at the suction chamber as a function of the operating condition.

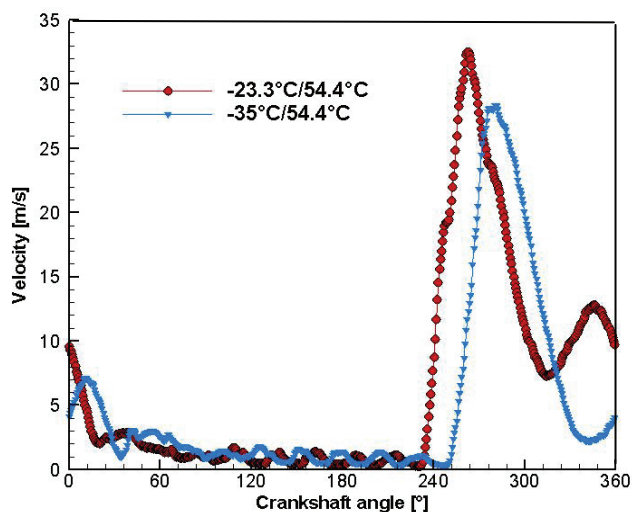


Figure 8: Velocity at the entrance of the suction chamber as a function of the operating condition.

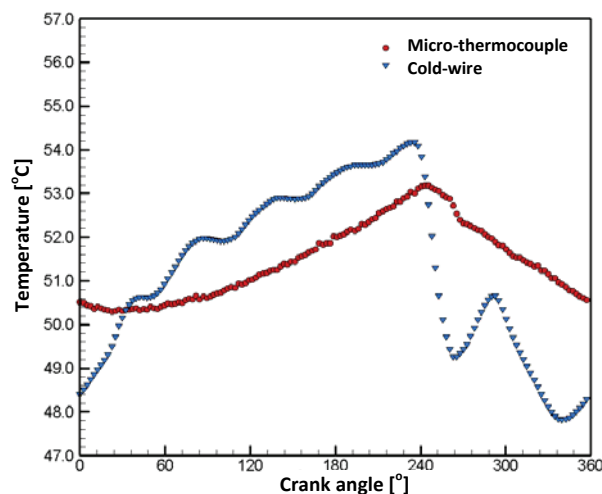


Figure 9: Measurements of temperature; operating condition -23.3°C/54.4°C.

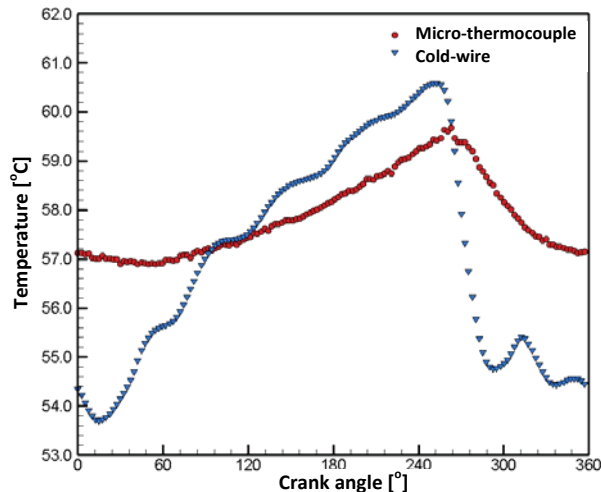


Figure 10: Measurements of temperature; operating condition -35°C/54.4°C.

#### 4. CONCLUSIONS

The present paper reported the results of an experimental investigation of thermal transients in the suction chamber of a high speed reciprocating compressor. To this extent, a cold-wire probe and a micro-thermocouple probe were specially designed for temperature measurements. Results for valve displacement and pressure pulsation in the suction chamber were also obtained to complement the analysis. The cold-wire sensor was seen to have a much faster response time than the micro-thermocouple and was able to measure steep transients of temperature and

velocity in the suction chamber. However, the cold-wire sensor is very fragile to handle and requires a very expensive and complex auxiliary system. Measurements indicated abrupt variations in all flow properties, strongly linked to the valve motion. When the valve is open, velocity reach levels of approximately 30m/s, in comparison with 3m/s during the period the valve is closed. It has been observed a considerable increase in the gas temperature during the period the suction valve is closed associated with heat transfer between the gas and the suction chamber walls.

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